CHAPTER 1
INTRODUCTION

1.1 Background of Study

The Shell Eco-Marathon is an educational project that challenges the student teams to design and build the most energy-efficient vehicle to compete against other teams' vehicles. The principle of the Shell Eco-Marathon is to design and build a vehicle that uses least amount of fuel to travel to the farthest distance.

Malaysia will become the host for the Asian programme for 3 consecutive years from 2010, thereafter it will be brought to another Asian country for the subsequent 3 years. For 2010 edition, the Shell Eco-marathon will be held on 8-10 July 2010 at the Sepang International Circuit in Kuala Lumpur, Malaysia.

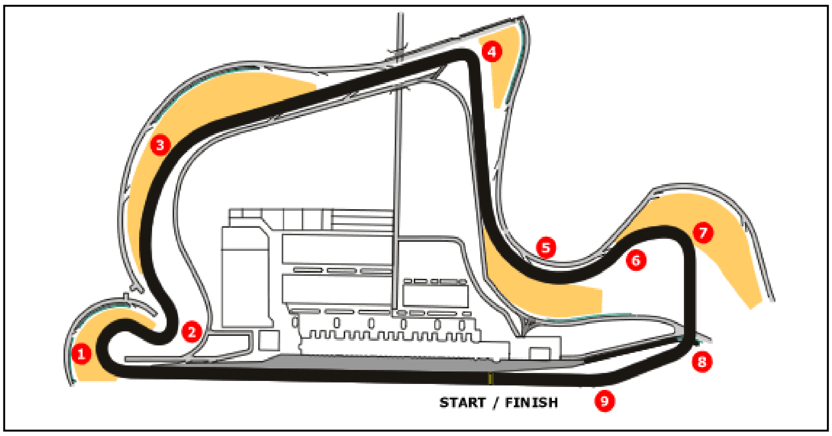


Figure 1.1: Sepang International Circuit where the race will take place.

In vehicle dynamic, some factors that influence fuel consumption of the vehicle are:

1. Engine performance
2. Drag force (effectively influence when vehicle at high speed)
3. Rolling force(vehicle at low speed)

Running chassis system is one of the most important parts in a vehicle to control the movement and provide ride comfort of the vehicle on the road. The parts include in this system are steering, brake, wheel and suspension. Running chassis system of vehicle will increase the usage of fuel when the rolling force between tire and road is high. The factor influences the rolling force are weight of the vehicle and the type of tire and wheel used. The vehicle with larger weight will has higher rolling force.

The project will be designed and selection of the steering system will based on the rule and regulation of the competition. The steering system also influences the ride comfort of the vehicle during driving especially on bad road condition. The precision of the steering also has to be considered in designing the steering system. After design and select the steering system that will be used, the students together with other group member have to fabricate the car. The car needs to be tested and analyzed to get the best performance.

1.2 Problem Statement

The Shell Eco-marathon challenges students around the world to design, build, and test vehicles that travel further using less energy. Steering system should be design as light and simple as possible in order to reduce the weight of the vehicle. The requirement of the Shell Eco-Marathon state the turning radius must be sufficient to enable safe overtaking as well as negotiating the curvature of the track. During cornering, the steering system determines whether the vehicle will have slipping or in case of oversteer or understeer.

The crude oil in the earth is depleting from time to time and the price of fuel also increases, this project is an innovation to design a vehicle that has low fuel consumption. Since Malaysia will be the host for Shell Eco-marathon for the first time in 2010, Universiti Teknologi PETRONAS has responsibility to ensure our country can compete with other team from other country especially in Asia region.

1.3 Objective

The objective of this project is to design and fabricate the lightest and simplest steering system in order to reduce the weight of the vehicle and meet the sufficient turning radius to be participated in Shell Eco-marathon 2010.

1.4 Scope of study

The project focuses on one part of running chassis which is steering system. The steering system has to be designed as light and simple as possible. The design should meets the competition rule which is the vehicle has sufficient turning radius to enable safe overtaking as well as negotiating the curvature of the track. The steering system is considered as an assembly consisting of steering wheel, steering column, steering shaft, steering gear box and steering linkages. After designing the steering system, the simulation has to be done in order to ensure it can work and some steering behaviors will be analyzed such as kinematic steering ratio and maximum front steer wheel angle. The complete steering system has to be fabricated and installed on the complete car. The car has to be tested whether it meets the requirement to participate in Shell Eco-Marathon 2010.

CHAPTER 2
LITERATURE REVIEW

The current European Shell Eco-marathon record for a combustion engine entry was set in 2004 by the team from Lycée La Joliverie (France) at 3,410 km on the equivalent of a single liter of fuel. For prototype vehicles using fuel cells, the record is even more impressive. In 2005, the hydrogen-powered vehicle built by Swiss team ETH Zurich achieved a projected 3,836 km on the equivalent of a single liter of fuel**.** This is the equivalent of driving from Paris to Moscow [1].

The top performing vehicles are specially designed for high efficiency. Some vehicles use a coast or burn technique whereby they briefly accelerate from 16 to 32km/h and then switch the engine off and coast for approximately 2 minutes until the speed drops back down to 16 km/h. This process is repeated resulting in average speed of 24km/h for the course [2]. Typically the vehicles have:

- Automobile drag coefficients (C_d) less than 0.1
- Rolling resistance coefficients less than 0.0015
- Weight without driver less than 45kg
- Engine efficiency less than 200 s.f.c. (cc/bhp/hr)

All the previous teams tried to fabricate the car including the steering system as light as possible. They use light material such as aluminium and carbon fiber to minimize the weight of the vehicle. The Dalhousie Supermileage team from Dalhousie University, Canada fabricated the running chassis components from 6061 T6 Aluminium round and square tubing [3]. Pac-Car ETH Zurich from Switzerland used advanced materials which are polyurethane foam and carbon-epoxy sandwich as their running chassis material [4].

For steering system, the previous team used simple link-joint mechanism and rack and pinion system. The example of teams was using simple link-joint mechanism are

Cal Poly Supermileage from California Polytechnic State University, USA[5], Dalhousie Supermileage[3] and National University of Singapore[6]. The example of team was using rack and pinion system is Northern Arizona University [7].

Figure 2.1: Rack and pinion system by Northern Arizona University

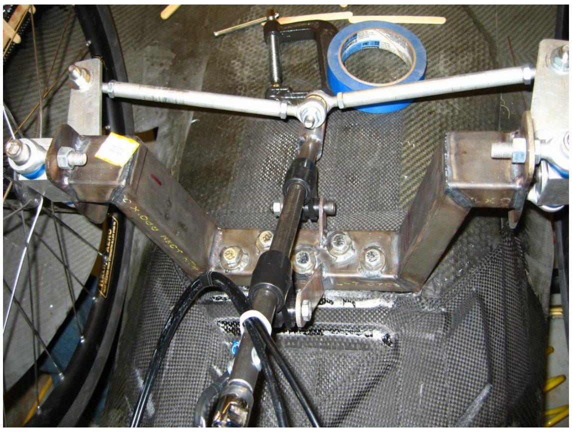


Figure 2.2: Simple link and joint mechanism by Cal Poly Supermileage

For steering wheel, the previous teams used joystick and handlebar. Most of the teams used handlebar as their steering wheel such as National University of Singapore, Cal Poly Supermileage and University of British Columbia[8]. Only Pac-Car from ETH Zurich used joystick as the steering wheel [4].

Most the previous teams were using front wheel steer. But there are some teams were using rear wheel steer which are Cedarville University[9] and Pac-CarII from ETH Zurich.

The Shell Eco-Marathon also become as an academic coursework for some academic institution such as National University of Singapore. The Shell Eco-Marathon is integrated into students' coursework as an optional module, again for third- and fourth-year students. The project is also a part of the studies programme at the Politecnico di Torino Italy, where time spent on the project is taken into consideration as part of students' internship requirements [6].

CHAPTER 3
THEORY

3.1 Ackermann Steering Geometry

The main conflicts entering into the design of steering systems concern the isolation from feedback of road shocks, while retaining sufficient road fell for positive control and striking an acceptable compromise between light steering efforts when parking and sufficient weight at speed. Much of the dynamic effects attributed by customers to the steering system are the result of suspension geometry and wheel control factors, especially in regard to toe settings and dynamic toe changes occurring at the front, rear or both [10].

The first approach to obtaining the best steering control and handling behavior might be to attempt to design the steering system so that the two front wheels run with equal slip angles during cornering. Initially, it might be assumed that the way to achieve this would be to design the steering system with Ackermann geometry as shown in figure 3.1 below. Ackermann steering geometry is a geometric arrangement of linkages in the steering of a vehicle designed to solve the problem of wheels on the inside and outside of a turn needing to trace out circles of different radii. However, the principle of Ackermann steering geometry assumes a vehicle cornering with its tires running at zero slip angles [10].

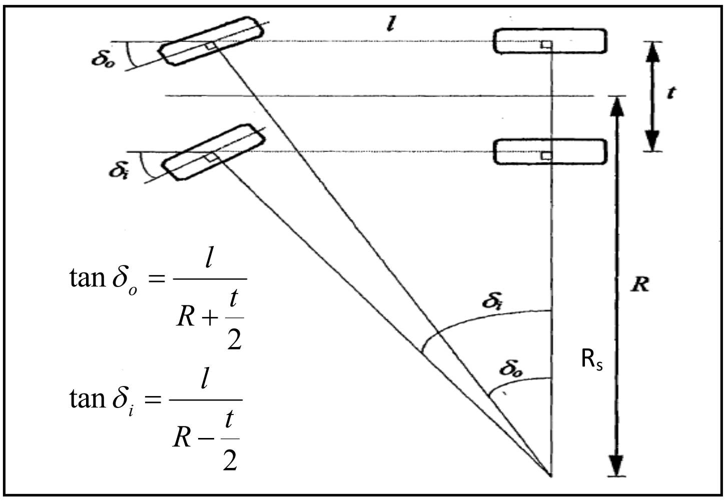


Figure 3.1: Ackermann steering geometry

In order to avoid skidding of tires when the vehicle is taking a turn, it is necessary that both the inner and outside wheel turn on arc which has a common centre of turn [11]. Slip angle as shown in Figure 3.2 below is the angular difference between the direction the tire contact patch with the road is pointing and the direction of the wheel.

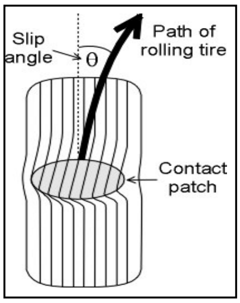


Figure 3.2: Slip angle

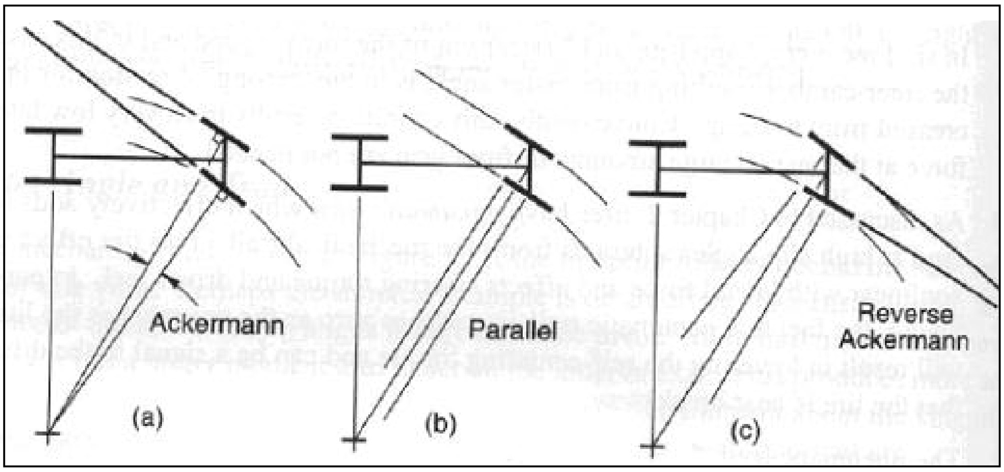


Figure 3.3: Variations of the Ackermann steering geometry

In ideal no slip situations, Ackermann steering would make perfect sense, but with issues such as slip angle and varying tire coefficients (due to temperature, vehicle speed, steering angle, etc), several variations of the Ackermann steering geometry have been used instead as shown in Figure 3.3 above. True Ackermann geometry would be a situation as seen by Figure 3.3 (a) where both wheels turn on arcs parallel to one another. As can been seen, the slip angle of the inner wheel is greater than the outside wheel. Parallel Ackermann steering is defined by having the same slip angle on both the inner and outside wheel (Figure 3.3 (b)). Reverse Ackermann is the exact opposite of true Ackermann geometry as the outside wheel is at a higher slip angle than the inner wheel.

At low speeds when the tires have minimal tire shear losses on dry, clean pavement, the true Ackermann steering geometry is beneficial as the tires are in almost a perfect situation of minute slip angle. Parallel or reverse Ackermann in this scenario would push (or understeer) the front of the car away from the desired path. In both situations, the inner tire contributes to this push similarly to a centrifugal force.

At high lateral accelerations, true Ackermann becomes disadvantageous as loads on the outside wheel increase and the greater slip angle of the inner tire creates higher tire temperatures and slows down the car due to tire drag. The inner tire has also surpassed the maximum slip angle of grip assuming the outside tire is already at the optimum slip angle. Parallel or reverse setups are more advantageous in this situation as both the inner and outside tires still have lateral grip. Reverse Ackermann steering can even be more beneficial than the parallel Ackermann geometry since the outside tire (which currently has more loads due to weight transfer) is at the optimum slip angle and the inner is at a lower slip angle with fewer grips. This in turn allows the inner tire to have grip but less than the outside tire, decreasing the effects of understeer [12].

During the development of a vehicle, the understeer or oversteer characteristics typically will be quantified in order to ensure the vehicle in safe condition during cornering. The original definition of understeer and oversteer was proposed by Olley[10], was based on relative sizes of front and rear slip angles. Understeer was defined as front wheel slip angle being larger than rear wheel and oversteer was defined as front wheel slip angles being smaller than rear wheel. The effect of slip angle on vehicle handlings that causes the understeer or oversteer is shown in Figure 3.4.

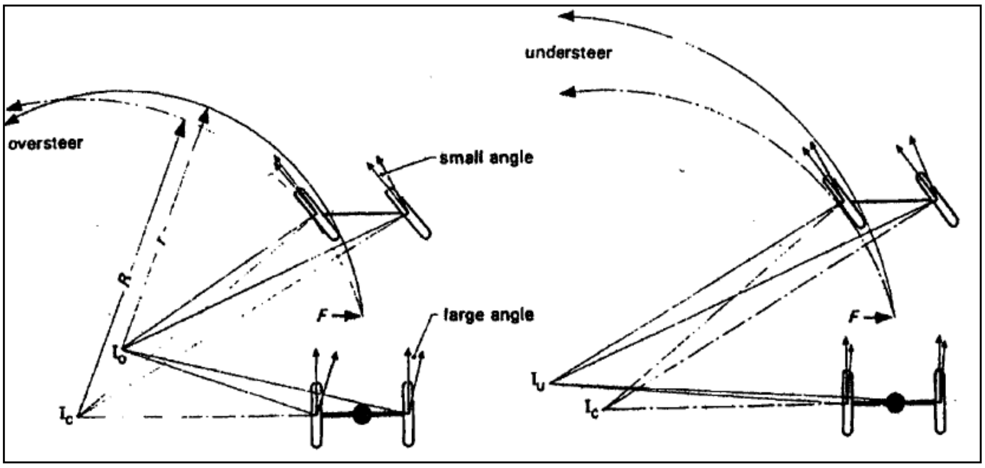


Figure 3.4: Effect of slip angle on vehicle handling

3.2 Steering Behaviour

The theoretical turning radius, R_s is the radius of circle which the outer wheel traces with the largest steering angle. The turning radius of vehicle should be as small as possible to make it easy to turn and park [13]. The formula:

$$R_s = \frac{l}{\sin\delta_o} + d \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.1)$$

Where δ_o is outer wheel angle, l is wheelbase and d is kingpin offset. Since kingpin offset is very small compared to wheelbase, the equation 3.1 can be written as:

$$R_s = \frac{l}{\sin\delta_o} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.2)$$

A typical passenger car turning radius is normally between 5.5m and 6.5m with SUV turning radius going out as much as 7.5m to 8.5m. In the record, the London taxi has a tiny 4m turning radius to allow it to do U-turns in the narrow London streets [14].

The kinematic steering ratio, i_s is one of the important characteristic for steering system that refers to the ratio of the steering wheel angle, δ_H to the mean steer angle, δ_m of a pair of steered wheels.

$$\delta_m = \frac{\delta_o + \delta_i}{2} \qquad \text{……………………….(3.3)}$$

$$i_s = \frac{\delta_H}{\delta_m} \qquad \text{……………………….(3.4)}$$

In general, the steering ratio of the vehicles varies from 13:1 to 16:1. For racing cars, the steering ratio is normally much smaller than for passenger cars that is closer to 1:1.This is because the racing drivers need to get fuller deflection into the steering as quickly as possible

In practice, a parallel steering-in of the front wheels (real layout) is targeted up to a steering angle of about 20° and only with larger steering angles, an approximation of the Ackermann condition is implemented [15].

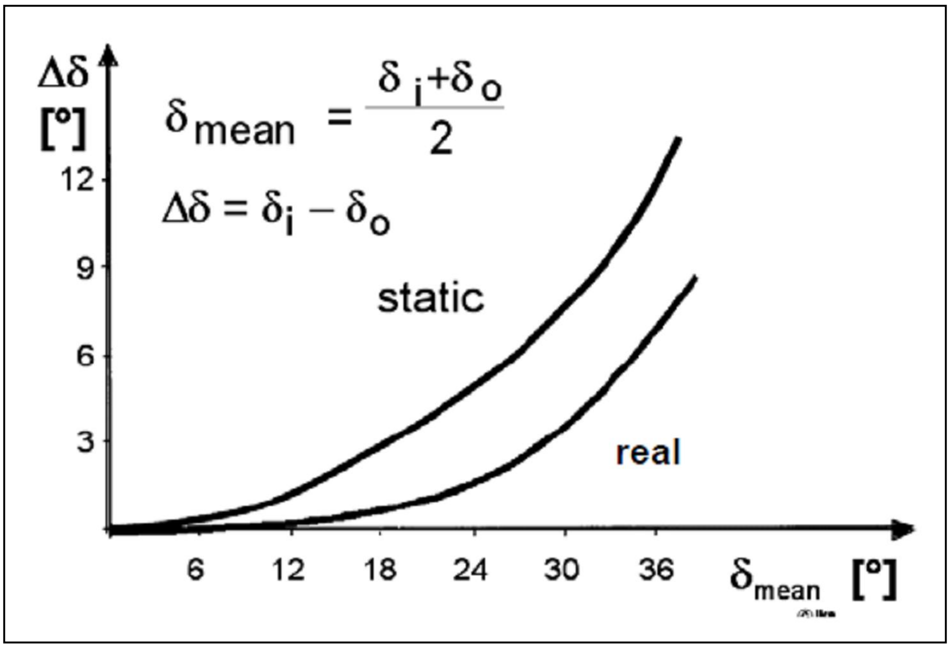


Figure 3.5: Toe-out angle as a function of the mean steering angle for static and real steering element layout

3.3 Self-Centering

The smooth operation of steering depends much upon the wheel alignment which means the vehicle has self centering [12]. The important alignment factors are caster angle, camber angle and kingpin inclination. The caster angle is the angle formed by the forward or backward tilt of the kingpin centre line from vertical when viewed from the side of the wheel. The purposes of caster angle are to maintain directional stability and control, to increase steering return ability and to reduce the driver's effort to turn the vehicle.

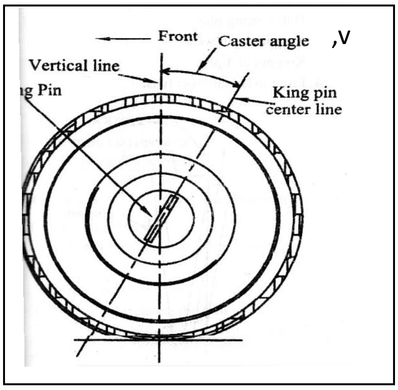


Figure 3.6: Caster angle

The chamber angle is the angle between the centre of the tire and the vertical line when viewed from the front. Kingpin inclination is the angle between the centre line of the kingpin and the vertical line. The purposes of kingpin inclination are to provide directional stability along with the caster angle and helps in self-centering of wheels after taking a turn.

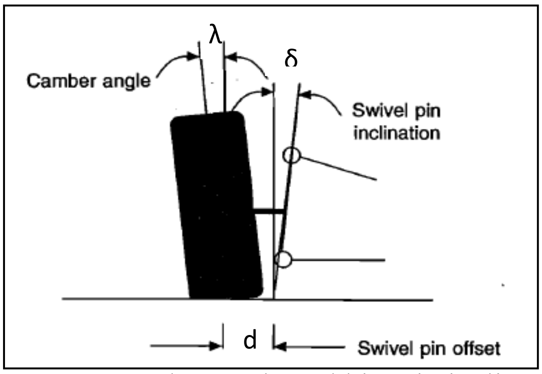


Figure 3.7: Camber angle and kingpin inclination

3.4 Mechanical Linkage

A mechanical linkage is a series of rigid links connected with joints to form a closed chain or a series of closed chains. Each link has two or more joints and the joints have various degrees of freedom(DOF) to allow motion between the links. A linkage is called a mechanism if two or more links are movable with respect to a fixed link. Mechanical linkages are usually designed to take an input and produce a different output, altering the motion, velocity, acceleration, and applying mechanical advantage.

The most common linkages have one degree of freedom(DOF), meaning that there is one input motion that produces one output motion. Most linkages are also planar, meaning all the motion takes place in one plane. Spatial linkages (non-planar) are more difficult to design and therefore not as common.
Kutzbach-Gruebler's equation is used to calculate the degrees of freedom of linkages. The number of degrees of freedom of a linkage is also called its mobility.

The general form of the Kutzbach-Gruebler equation for planar linkages involving more complex joints:

$$m = 3(n - j - 1) + \sum_{n=1}^{j} f_i, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.5)$$

and for spatial linkages (linkages involving 3D motion):

$$m = 6(n - j - 1) + \sum_{n=1}^{j} f_i, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.6)$$

Where:

m= mobility (degrees of freedom)

n= number of links (including a single ground link)

j= number of total joints, regardless of connectivity or degree-of-freedom

$\sum_{n=1}^{j} f_i$ = sum of each joint's individual degree of freedom

Mechanisms with one degree of freedom are termed constrained mechanisms. As mentioned, most mechanisms used in machines are constrained. Mechanism with zero, or negative degree of freedom are termed locked mechanisms. These mechanisms are unable to move and form a structure. Mechanisms with more than one degree of freedom are termed unconstrained mechanisms. These mechanisms need more than one driver to precisely operate them [16].

CHAPTER 4
METHADOLOGY

4.1 Overview

The methodology is formulated based on Morris Asimow's morphology of design. According to the scope of project, project activities will start from problem definition until design communication. Final output of this project is detail design for prototype fabrication. Fabrication of fully-working prototype is out of the project scope.
Project timeline is shown in the Gantt Chart attached in Appendix . Detail methodology is shown in Figure 4.1 and 4.2.

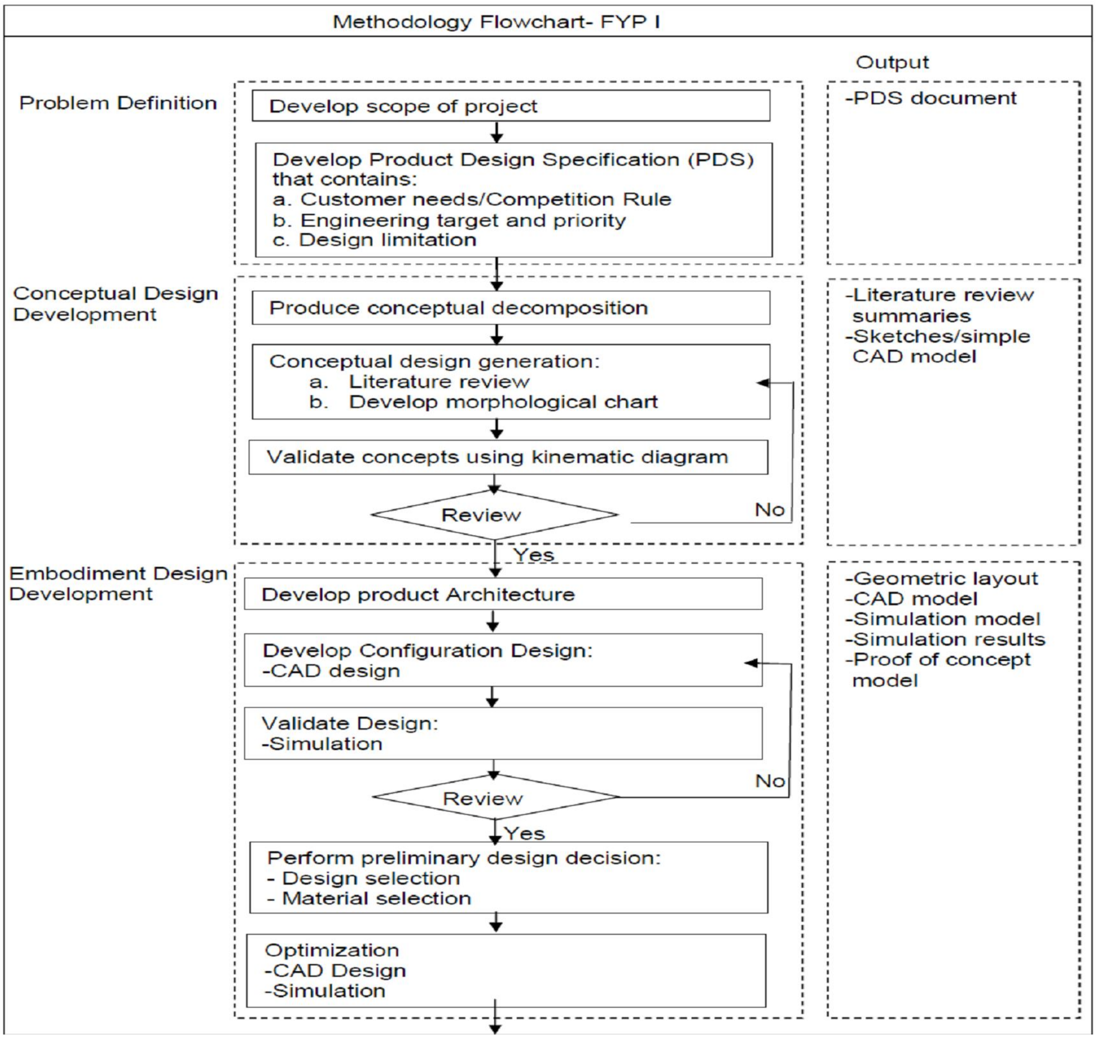


Figure 4.1: Flow chart of methodology FYP I

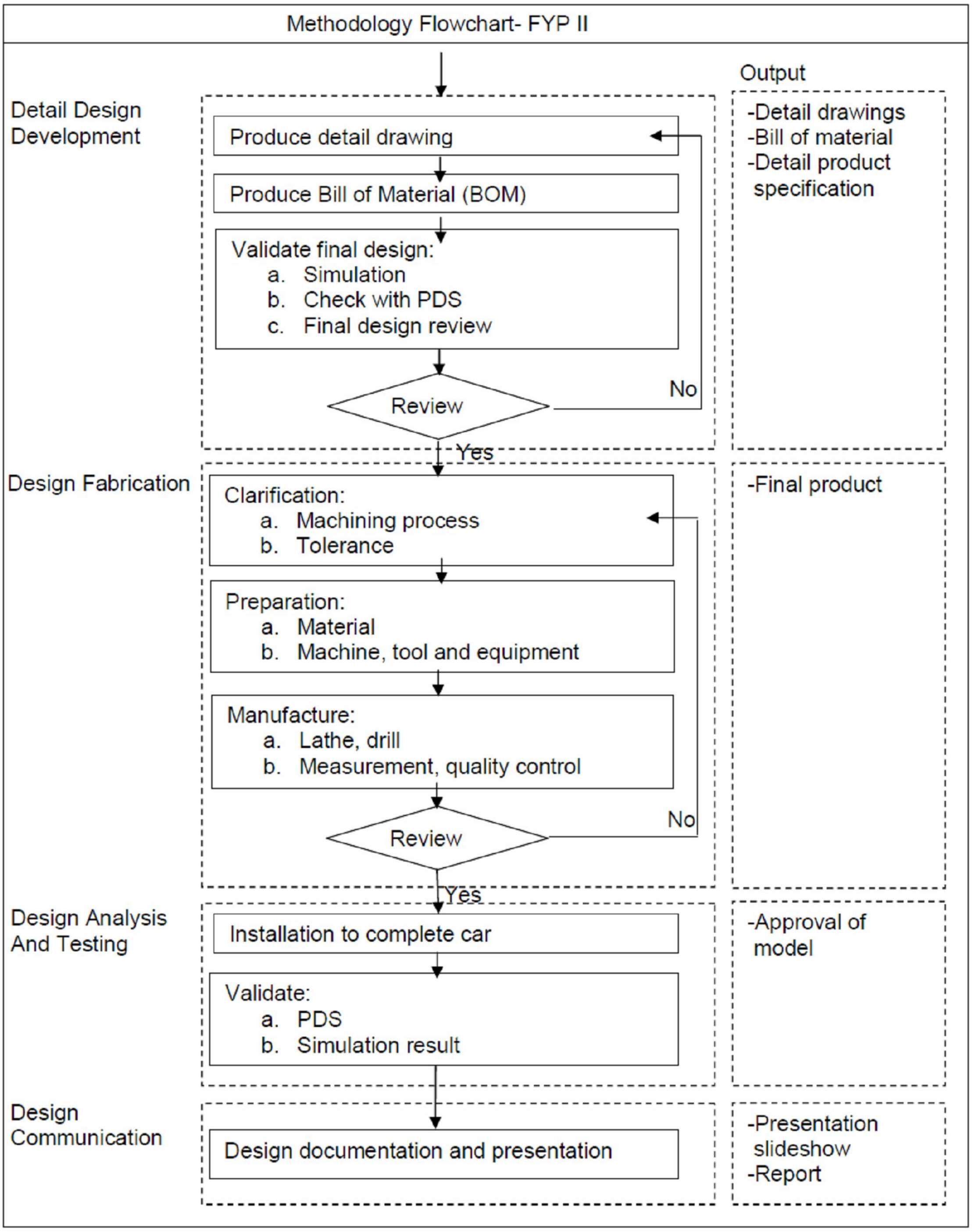


Figure 4.2: Flow chart of methodology FYP II

As shown in the flow chart of methodology in figure 4.1 and 4.2, Table 4.1 below indicates the expected output for each design phase:

Table 4.1: Expected output for each design phase

Project Phase	**Expected Output**
Problem Definition	• PDS document
Conceptual Design	• Literature review summaries • Sketches / simple CAD models
Embodiment Design Development	• Geometric layout • CAD model • Simulation model • Simulation results • Proof of concept model
Detail Design Development	• Detail drawings • Bill of materials(BOM) • Detailed product specification
Design Fabrication	• Final product
Design Analysis and Testing	• Approval of model
Design Communication	• Presentation slideshow • Report

4.2 Problem Identification

Understand the principle of Shell Eco-marathon and the rule and regulation about steering system to participate in the competition. Identify the objective, problem statement, job scope and the significant of the project to the student and university.

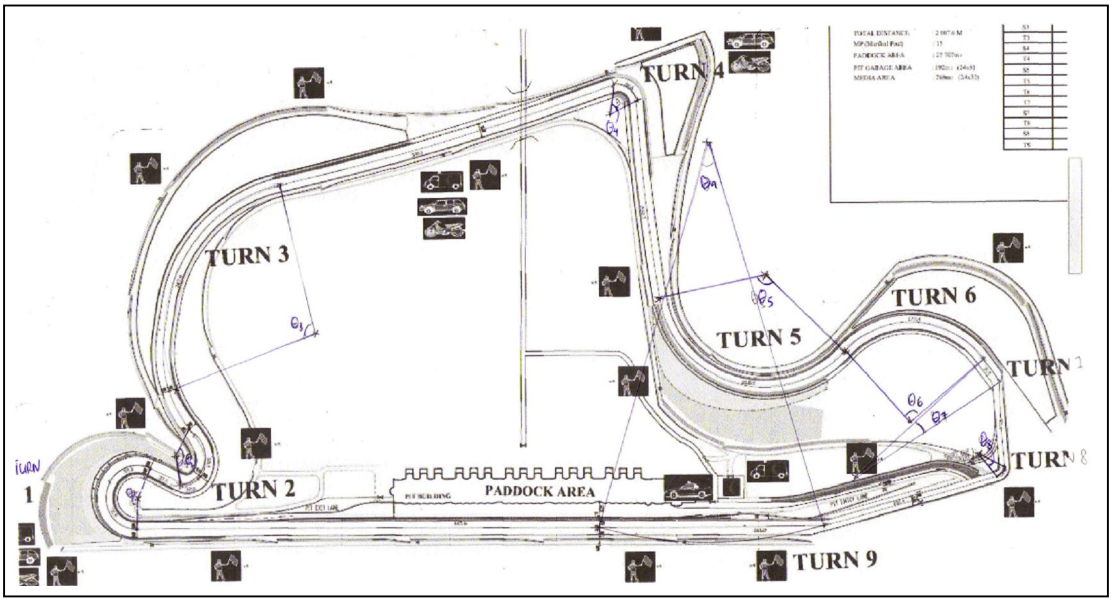


Figure 4.3: Measuring the radius of turn in Sepang circuit

All the turn at Sepang circuit has been analyzed to know the radius in order to ensure the steering system will be designed has sufficient turning radius. The data is tabulated in table 4.2 below.

Table 4.2: Detail of turn in Sepang Circuit

Turn	Length(m)	Angle	Radius(m)
1	125.2	200^o	35.87
2	88.2	150^o	33.70
3	347.9	97^o	205.50
4	36.6	60^o	34.95
5	268.8	123^o	125.21
6	177.4	90^o	112.94
7	31.5	10^o	180.48
8	13.8	25^o	31.63
9	166.0	32^o	297.22

Based on the table 4.2 above, the smallest radius is approximately about 31.63m at turn number 8.

The output of problem definition process is a control document named as Product Design Specification (PDS). The PDS for the steering system is as follows:
General Features:

a. It should be light weight
b. Steering geometry should not get affect by bad road conditions
c. It must be easily operateable
d. Minimum space needed

For the purpose of this project, the specific requirements are identified in order to achieve the objective of this project. The requirements are:

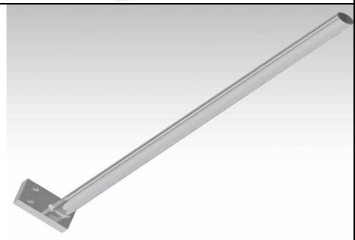
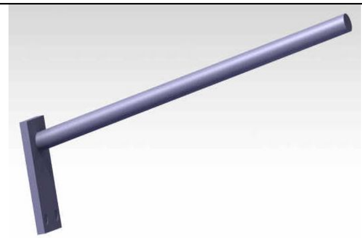
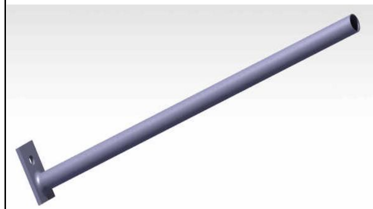
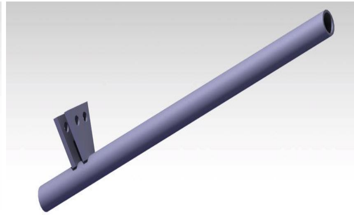
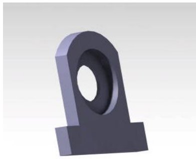
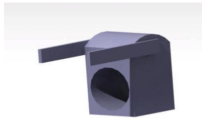
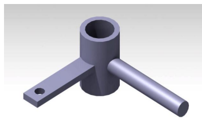
a. Front wheel steer
b. Turning radius need to be smaller than radius of turn in Sepang circuit: 4.0-6.5m
c. Steering ratio is closer 1:1
d. Maximum angle for outer wheel 30^o. This value will be ensured the car will has turning radius between 4.0m. See Appendix for calculation.
e. Follow the Ackermann steering geometry.
f. The difference between angle for inner wheel and outer wheel small as possible
g. Light to turn the wheel(kingpin inclination angle, castor angle and camber angle is equal to zero)

4.3 Conceptual Design

Before producing the design concepts, the research was conducted to find out about other teams that have been participated in Shell Eco-marathon and their design. The research was conducted by reading and searching the previous teams and organizers' website. The next of step of this phase is to produce design concepts that would perform as required.

The morphological chart was used to uncover combination of ideas that comprise design concept that might not normally be generated.

Table 4.3: Morphological chart for design concept generation

Part	Function	Concept 1	Concept2	Concept 3	Concept 4
Steering wheel	-Steer input				
Steering column	-Tansmit steer input				
Steering column holder	-Hold the steering cloumn				
Tie rod	-Connect steering column to spindle				
Spindle	-Steer the wheel				

By combining concepts from the morphological chart, four (4) conceptual designs are generated as follows:

Design 1

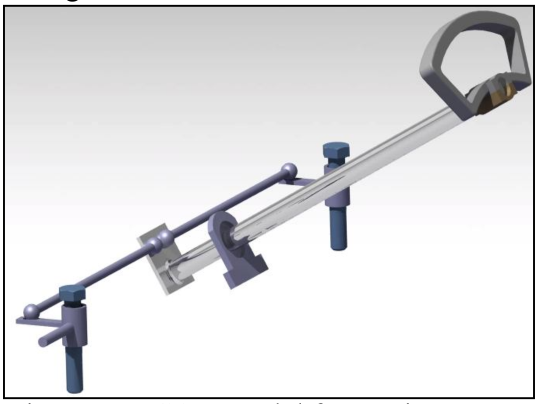


Figure 4.4: CAD model for Design 1

Design 2

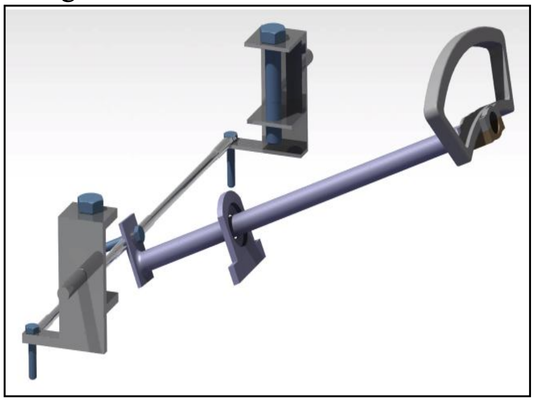


Figure 4.5: CAD model for Design 2

Design 3

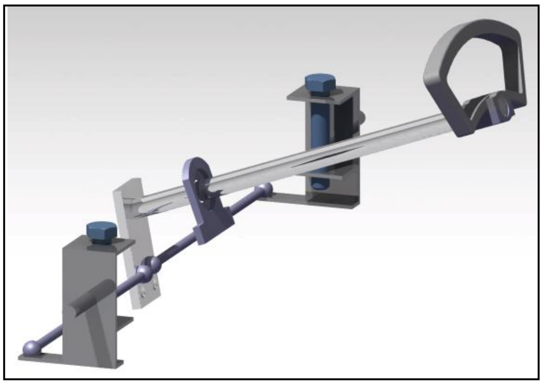


Figure 4.6: CAD model for Design 3

Design 4

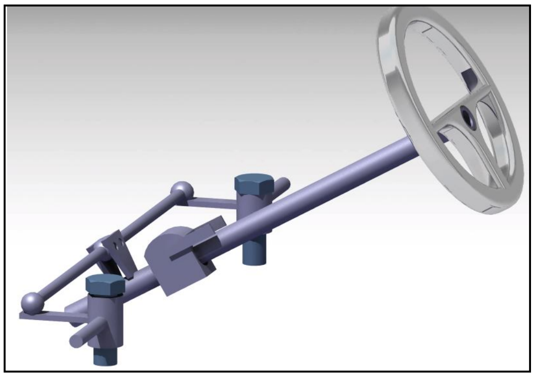


Figure 4.7: CAD model for Design 4

4.4 Embodiment Design Development

Use CATIA software to design the conceptual steering system. Overall four alternatives steering system has been designed. Next, all the alternatives steering system was simulated by using ADAMS software to analyze the mechanism and the steering behavior. Based on the result obtained from ADAMS, only one alternative steering system was selected by using weighted property index method to be used in the competition. The design was selected has been optimized to increase the maximum angle for inner and outer wheel in order to ensure the sufficient of turning radius. There are three options of materials that can be used in fabricating all parts of steering system which are aluminium, carbon fiber and steel and only one material was selected by using weighted property index method.

4.5 Detail Design Development

Provide the detail drawing for selected steering system by using CATIA software. The finalized drawing was simulated in ADAMS and the result has been reviewed in order to ensure it is follow the Product Design Specification(PDS).

4.6 Design Fabrication

Fabricate the steering components together with the frame of the car. Tolerance is the important criteria in machining process in order to ensure that the steering components can be assembled with other part of the car.

4.7 Design Analysis and Testing

Test the steering system to control the movement of the complete vehicle. The problems rose from the steering system such as mobility, convenience and workability has been analyzed. The comparison between the actual results with the simulation result has been made.

CHAPTER 5
RESULTS AND DISCUSSION

5.1 Design Development

The basic of the steering system as shown in Figure 5.1 below:

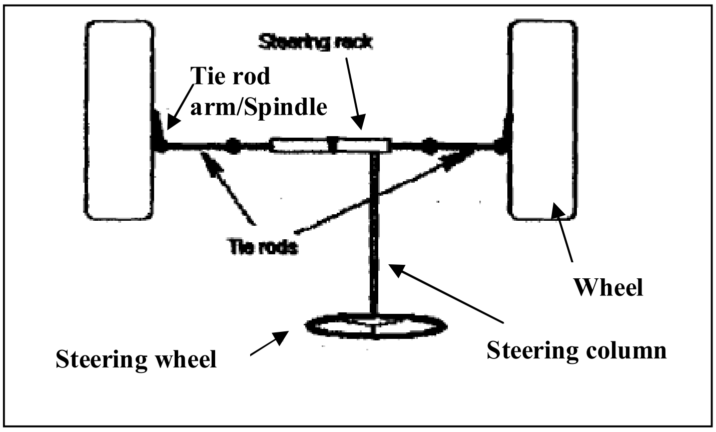
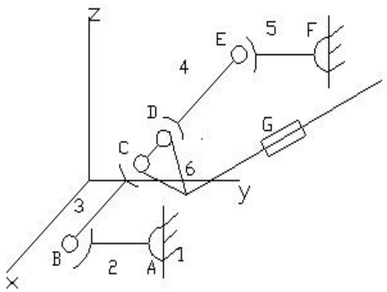
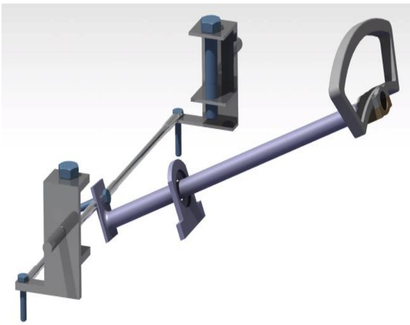
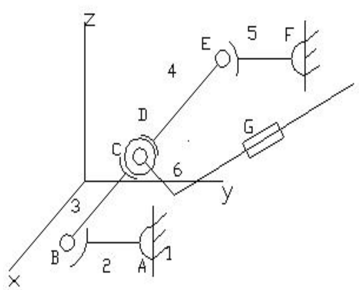


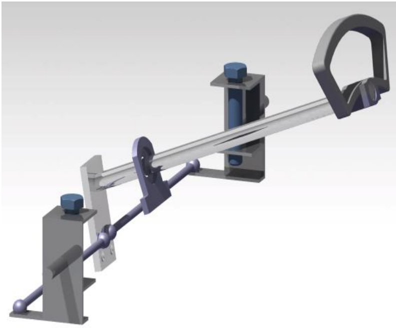
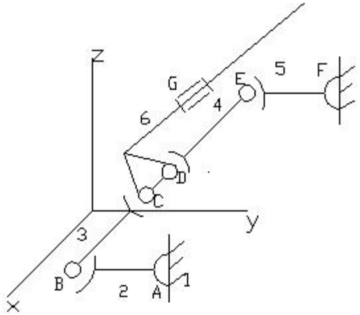
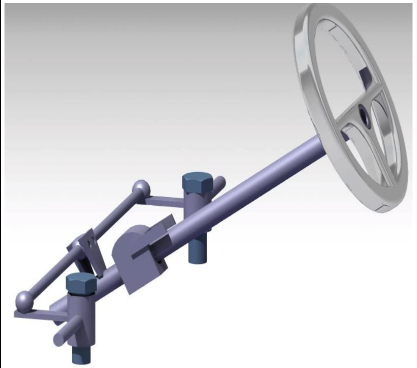
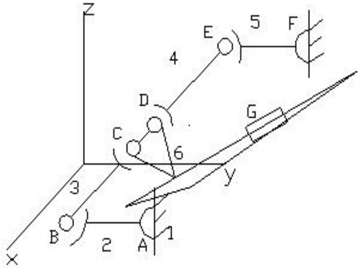
Figure 5.1: Basic steering system

Base on the Figure 5.1 above, the parts of steering system includes steering wheel, steering column, tie rod, steering rack, tie rod arm and spindle.

Four designs have been proposed in the beginning. After make some analysis, simulation and comparison between all the designs, only one design will be selected. Table 5.1 shows the designs have been proposed.

Table 5.1: Designs were proposed in the beginning for steering system

Design	CAD Drawing	Kinematic Drawing	Validate	Description
1			The mechanism has six links(n=6), three rotational joints(A, F, G) and four spherical joint(B, C, D, E). The number of DOF for this mechanism: $m = 6(n - j - 1) + \sum_{n=1}^{j} f_i$ $= 6(6 - 7 - 1) + (3\times 1 + 4\times 3)$ $= 3$ Since DOF >0, the mechanisms is unconstrained.	Use simple link mechanism. Four ball joint is used to connect steering column with spindle.
2			The mechanism has six links(n=6), three rotational joints(A, F, G) and four spherical joint(B, C, D, E which C and D is positioned at same place). The number of DOF for this mechanism: $m = 6(n - j - 1) + \sum_{n=1}^{j} f_i$ $= 6(6 - 7 - 1) + (3\times 1 + 4\times 3)$ $= 3$ Since DOF >0, the mechanisms is unconstrained.	Use simple link mechanism. The mechanism almost similar with go-kart steering system. The tie rod is connected directly to hub axle by using ball joint.

3			The mechanism has six links(n=6), three rotational joints(A, F, G) and four spherical joint(B, C, D, E). The number of DOF for this mechanism: $m=6(n-j-1)+\sum_{n=1}^{j} f_i$ $=6(6-7-1)+(3\times1+4\times3)$ $=3$ Since DOF >0, the mechanisms is unconstrained.	Use simple link mechanism. Slightly similar with design1. But the connection position of the tie rod below the steering column.
4			The mechanism in has six links(n=6), three rotational joints(A, F, G) and four spherical joint(B, C, D, E). The number of DOF for this mechanism is $m=6(n-j-1)+\sum_{n=1}^{j} f_i$ $=6(6-7-1)+(3\times1+4\times3)$ $=3$ Since DOF >0, the mechanisms is unconstrained.	Use simple link mechanism. Four ball joint is used to connect steering column with spindle. Most of go-kart use this design.

5.2 Simulation

All the designs have been analyzed by using ADAMS to ensure that the design can function well. The motion was set in the steering column to simulate the mechanism. The simulation will show whether the design has functionally worked or not. By using ADAMS Postprocessor, the data of simulation can be viewed in graph and worksheet.

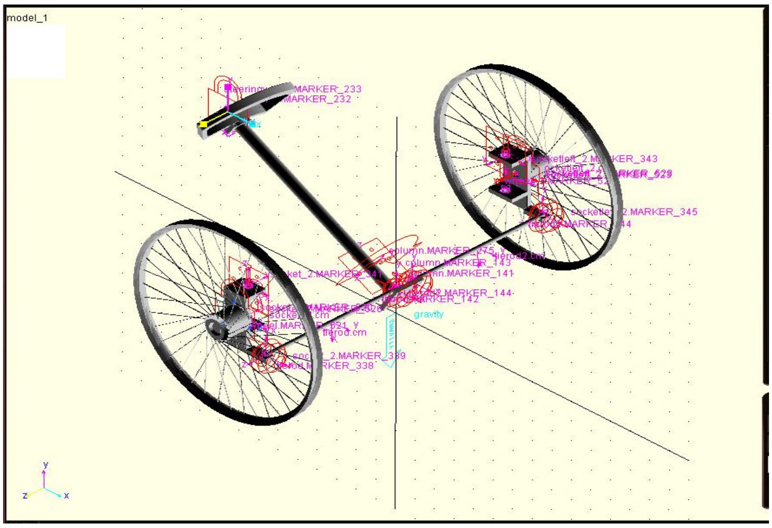


Figure 5.2: Simulation of the design for steering system in ADAMS view

5.2.1 Simulation Result

Design1

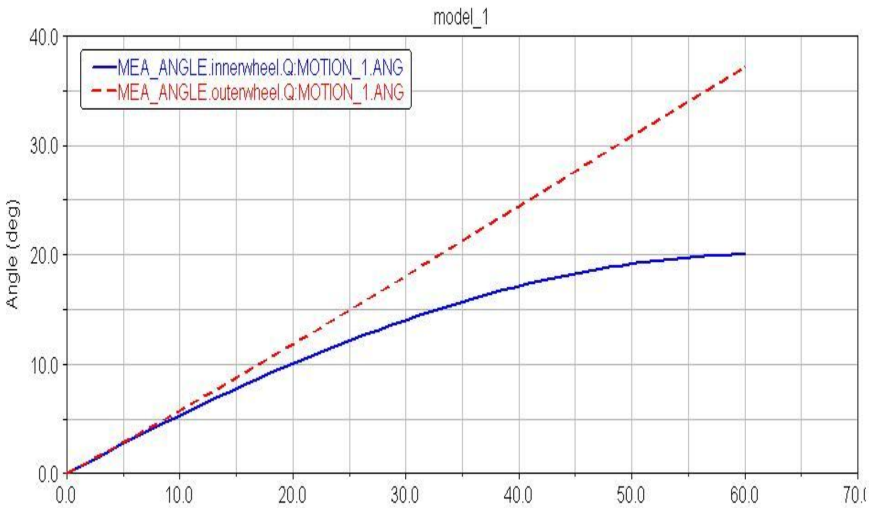


Figure 5.3: Graph of the front steer wheel angle versus steering wheel angle for design 1

Design 2

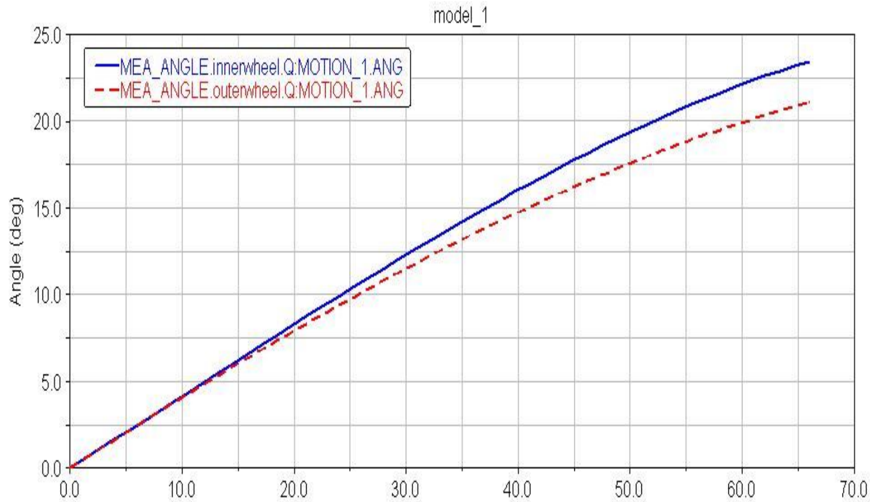


Figure 5.4: Graph of the front steer wheel angle versus steering wheel angle for design 2

Design 3

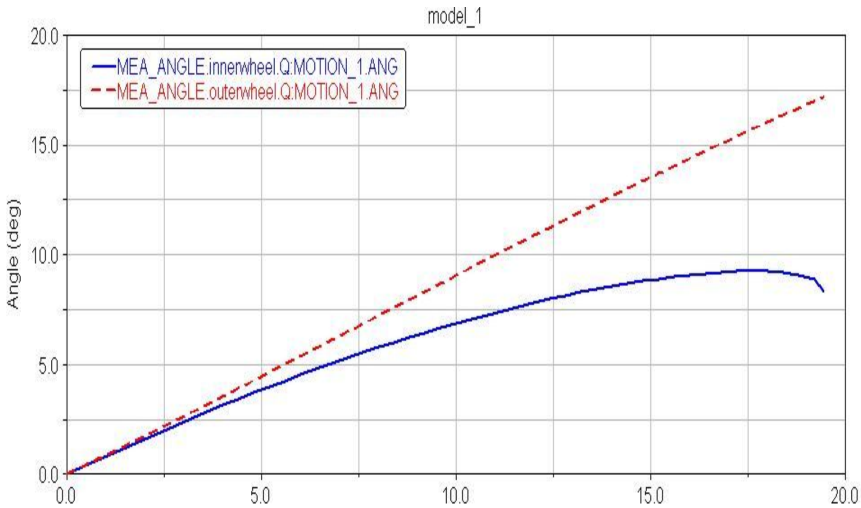


Figure 5.5: Graph of the front steer wheel angle versus steering wheel angle for design 3

Design 4

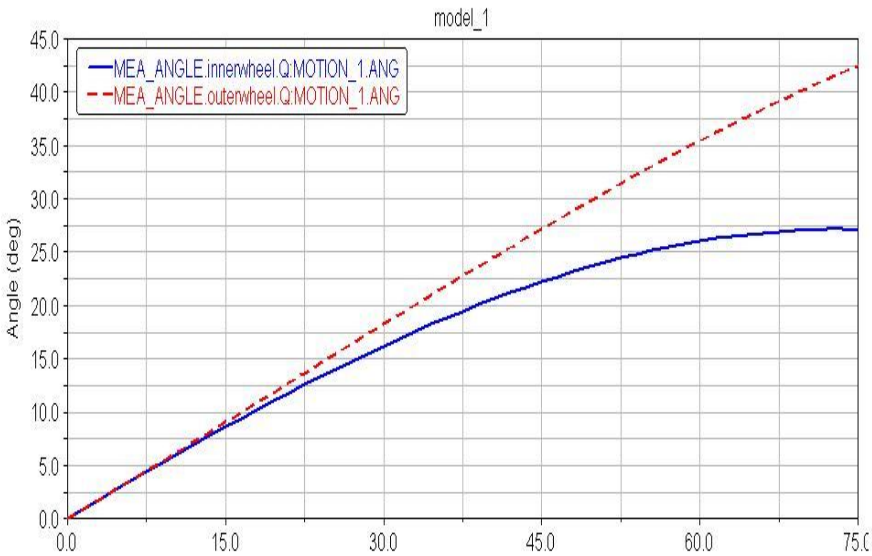


Figure 5.6: Graph of the front steer wheel angle versus steering wheel angle for design 4

The result of ADAMS simulation for all design is tabulated in Table 5.2 below:

Table 5.2: Result of ADAMS simulation for all design

Design	**1**	**2**	**3**	**4**
Angle outer wheel, δ_o	30	30	30	30
Inner wheel, δ_i	18.8	Disable	Disable	23.6
Steering wheel angle, δ_H	48	Disable	Disable	49.5
$\lvert\delta_i - \delta_o\rvert$	11.2^o	-	-	6.4^o
Steering ratio	1.97	-	-	2.00
Turning radius(m)	4.0	-	-	4.0
Direction	Same orientation with steering wheel	Same orientation with steering wheel	Opposite orientation of steering wheel	Same orientation with steering wheel
Ackermann	Reverse	True	Reverse	Reverse

5.2.2 Selection of The Steering System

Design Selection by using weighted property index and weight factor for steering design selection as shown in Table 5.3 below.

Table 5.3: Weight Factor for steering design selection

Criteria	**Weight factor**
Functionality	0.40
Steering ratio	0.25
Difference between angle for inner wheel and outer wheel, $|\delta_i - \delta_o|$	0.15
Ackermann geometry	0.10
Space	0.10

The weight factor is determined based on Product Design Specification(PDS). The highest weight factor is functionality, followed by the steering ratio, the difference between angle for inner wheel and outer wheel, the type of Ackermann geometry and the lowest weight factor is the space required.

Table 5.4: Weight property index for steering design selection

	Design 1		**Design 2**		**Design 3**		**Design 4**	
	Score	**Rating**	**Score**	**Rating**	**Score**	**Rating**	**Score**	**Rating**
Functionality	8	3.20	5	2.00	3	1.20	8	3.20
Steering ratio	9	2.25	2	0.50	2	0.50	8	2.00
Difference between angle for inner wheel and outer wheel, $|\delta_i - \delta_o|$	8	1.20	2	0.30	2	0.30	9	1.35
Ackermann geometry	7	0.70	9	0.90	7	0.70	7	0.70
Space	9	0.90	8	0.80	8	0.80	7	0.70
Weight property index	8.25		4.50		3.50		7.95	

The weighted property matrix as shown in Table 5.4 is applied to compare the design 1, 2, 3 and 4. Since the Table 5.4 shows the design 1 has the highest weight property index, it will be chosen. After the selection of the steering system design, the optimizations of the steering system have been done to ensure that the difference between angle for inner wheel and outer wheel is smaller as possible and the steering ratio is closer to 1.

5.3.Material Selection

Material selection is important in order to produce the new product. There are many materials in this world and every materials exist has their own properties and characteristic. In general, each new product is invented to satisfy the functions they need.

In order for the steering system to be light as possible and be able to perform well, several properties of the steering system's part have to be considered. In the selected design, the steering system consists of three main parts which are steering column, tie rod and spindle. The properties which will be needed in the three main parts are as below:

Table 5.5: Properties for all steering system parts must have

Part	**Steering Column**	**Tie Rod**	**Spindle**
Properties	• Lightweight • Can absorb a great amount of impact • Machinable • Low cost • Long lifetime	• Lightweight • Can absorb a great amount of impact • Machinable • Low cost • Long lifetime	• Lightweight but strong and sturdy • Have high value of hardness • Weldable • Machinable • Long lifetime and durable • Able to absorb huge pressure and vibrations • Low cost

With all the properties listed out, the further research is conducted on the materials which will be used to produce all the parts. In order to shorten the list of materials, several mechanical properties have to be met as shown in Table 5.6 below.

Table 5.6: Essential mechanical properties that the materials must have

Part	**Steering Column**	**Tie Rod**	**Spindle**
Mechanical Properties	• Low density • High modulus of elasticity • Non-corrosive • High ductility • High toughness	• Low density • High modulus of elasticity • Non-corrosive • High ductility • High toughness	• Low density • High modulus of elasticity • High ductility • High toughness • Weldability • Non-corrosive

Based on several aspects and observations, the best three materials that can be used to produced all the parts have narrowed down. The materials with its mechanical properties are mentioned in the Table 5.7 below.

Table 5.7: Mechanical properties for Carbon Fiber, Aluminium Alloy and Stainless Steel

Material	**Carbon Fiber**	**Aluminium Alloy**	**Stainless Steel**
Density (kg/m^3)	1780	2600-2800	6920-9130
Elastic Modulus(GPa)	230	69-79	190-200
Yield Strength(MPa)	-	35-124	107-415
Tensile Strength(MPa)	2500-4500	90-228	415-725
Ductility(%elongation)	0.6-2.0	6-40	5-40
Fracture Toughness(MPa√m)	1.6-9.0	24-44	76
Machinability	Very abrasive, difficult to machine. Require careful handling and removal of debris to avoid	Very easy to machine, tend to has poor surface finish	Depend on ductility and hardness. If too ductile, chip formation can produce built-up

	contact with inhaling of the fibers		edge, leading to poor surface finish, if too hard, it can cause abrasive wear of the tool
Weldability	Cannot be welded, can be joined to other parts by using screw.	Need special skill and equipment to be welded	Easy to be welded
Price	Expensive	Moderate	Cheap

5.3.1 Weighted Decision Matrix for steering column and tie rod

Since the properties which will be needed to produce steering column and tie rod almost same, the material will be used is same.

Table 5.8: Weight property index for material selection for steering column and tie rod

		Carbon Fiber		**Aluminium Alloy**		**Stainless Steel**	
Factor	**Weight**	**Score**	**Rating**	**Score**	**Rating**	**Score**	**Rating**
Density	0.30	9	2.70	8	2.40	7	2.10
Machinable	0.20	5	1.00	9	1.80	8	1.60
Non-corrosive	0.15	9	1.35	9	1.35	3	0.45
Ductility	0.15	9	1.35	7	1.05	8	1.20
Toughness	0.15	4	0.60	7	1.05	8	1.20
Cost	0.05	2	0.10	8	0.40	9	0.45
		7.10		8.05		7.00	

The weighted property matrix is applied to compare the materials which are carbon fiber, aluminium alloy and stainless steel that is possible to be used for fabricating steering column and tie rod. Since the table 5.8 shows that aluminium alloy has the highest weight property index, aluminium alloy will be chosen as the material for the steering column and tie rod.

5.3.2 Weighted Decision Matrix for spindle

Table 5.9: Weight property index for material selection for spindle

		Carbon Fiber		**Aluminium Alloy**		**Stainless Steel**	
Factor	**Weight**	**Score**	**Rating**	**Score**	**Rating**	**Score**	**Rating**
Density	0.25	9	2.25	8	2.00	7	1.75
Ductility	0.20	9	1.80	7	1.40	8	1.60
Toughness	0.20	9	1.80	7	1.80	8	1.60
Weldable	0.10	0	0.00	5	0.50	9	0.90
Machinable	0.10	5	0.50	9	0.50	8	0.80
Non-corrosive	0.10	9	0.90	9	0.90	6	0.60
Cost	0.05	2	0.20	8	0.40	9	0.45
Weight property index		7.45		7.50		7.60	

The weighted property matrix is applied to compare the materials which are carbon fiber, aluminium alloy and stainless steel that is possible to be used for fabricating spindle. Since the Table 5.9 shown that stainless steel has the highest weight property index, stainless steel will be chosen as the material for the spindle.

5.4 Optimization

Design 1 will be optimized in order to ensure that the steering system meet the product design specification (PDS) which are steering ratio is closer to 1 and the difference between angle for inner wheel and outer wheel small as possible

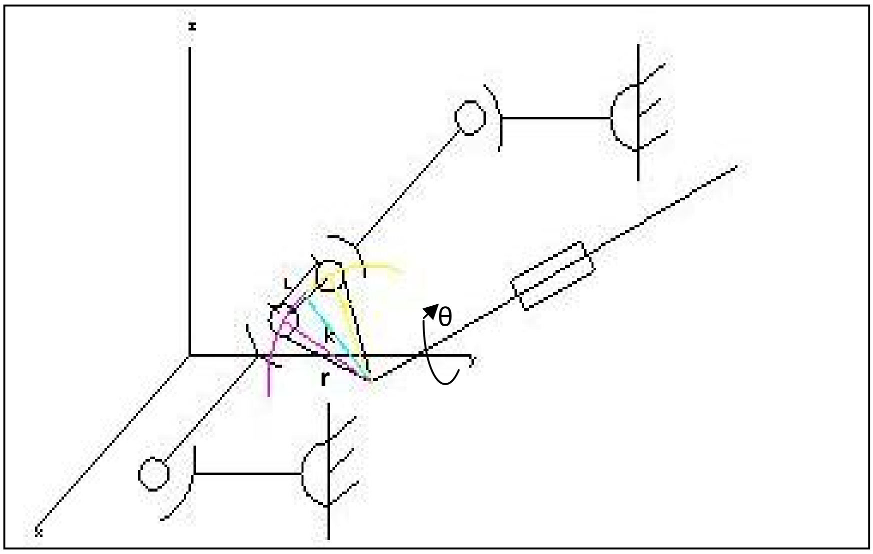


Figure 5.7: Optimization of the design

The equation for arc length, $s=r\theta$ ……………….(5.1)

Where r is radius and θ is angle in radian. By varying r, the arc length will change. Hence position of the tie rod also changes.

From Figure 5.7: $r=\sqrt{k^2+\left(\frac{L}{2}\right)^2}$ …...………...(5.2)

Where k is perpendicular distance between tie rod with axis of steering column's rotation and L is distance between two tie rods. In order to meet the requirements, some value for k and L were choose in designing the steering column and will be going to simulation.

Table 5.10: The result of ADAMS simulation for various value of k and L

k(mm)	**L(mm)**	**Outer angle, δ_o**	**Inner angle, δ_i**	**Steering wheel angle, δ_H**	**Difference between angle for inner wheel and outer wheel, $\lvert\delta_i - \delta_o\rvert$**	**Steering ratio, i**
35	30	30.0°	18.9°	48.0°	11.1°	1.963190184
35	35	30.0°	18.9°	47.6°	11.1°	1.946830266
35	40	30.0°	17.7°	47.0°	12.3°	1.970649895
35	45	30.0°	17.0°	47.0°	13.0°	2.000000000
35	50	30.0°	15.0°	44.0°	15.0°	1.955555556
40	30	30.0°	21.0°	41.2°	9.0°	1.615686275
40	35	30.0°	19.8°	41.0°	10.2°	1.646586345
40	40	30.0°	19.2°	40.5°	10.8°	1.646341463
40	45	30.0°	18.1°	40.0°	11.9°	1.663201663
40	50	30.0°	17.5°	40.0°	12.5°	1.684210526
45	30	30.0°	23.0°	40.0°	7.0°	1.509433962
45	35	30.0°	22.3°	38.0°	7.7°	1.453154876
45	40	30.0°	21.6°	37.3°	8.4°	1.445736434
45	45	30.0°	21.0°	37.3°	9.0°	1.462745098
45	50	30.0°	20.4°	37.2°	9.6°	1.476190476
50	30	30.0°	24.2°	35.0°	5.8°	1.291512915
50	35	30.0°	23.8°	34.5°	6.2°	1.282527881
50	40	30.0°	22.3°	33.6°	7.7°	1.284894837
50	45	30.0°	21.5°	33.6°	8.5°	1.304854369
50	50	30.0°	20.8°	32.5°	9.2°	1.279527559
55	30	30.0°	24.7°	31.8°	5.3°	1.162705667
55	35	30.0°	24.2°	31.6°	5.8°	1.166051661
55	40	30.0°	23.8°	31.2°	6.2°	1.159851301
55	45	30.0°	23.5°	31.0°	6.5°	1.158878505
55	50	30.0°	23.0°	31.0°	7.0°	1.169811321

From the Table 5.10 above, by increasing the k value with maintaining the value of L, the maximum inner angle is increasing. But when increasing L value with maintaining the value of k, the maximum inner angle is decreasing. By changing the k and L value, it will affect the difference between angle for inner wheel and outer wheel and also steering ratio. Based on the table above, the best value of k is 55mm and L is 30mm.

The detail result of design 1 with k=55mm and L=30mm is shown in Figure 5.8 and Table 5.11 below.

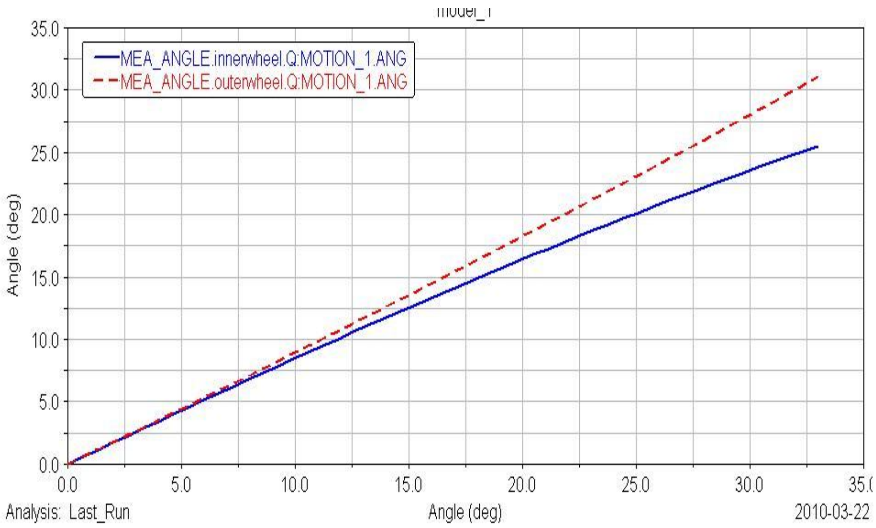


Figure 5.8: Graph of the front steer wheel angle versus steering wheel angle for design 1 with k=55mm and L=30mm

Table 5.11: The simulation result for design 1 with k=55mm and L=30mm

Steering wheel angle, δ_H	Outer angle, δ_o	Inner angle, δ_i	Mean wheel angle. δ_{mean}	Difference between angle for inner wheel and outer wheel, $\lvert\delta_i - \delta_o\rvert$
0^o	0^o	0^o	0^o	0^o
5^o	4.4^o	4.4^o	4.40^o	0^o
10^o	9.4^o	9.2^o	9.30^o	0.2^o
15^o	13.6^o	12.6^o	13.10^o	1.0^o
20^o	18.8^o	16.8^o	17.80^o	2.0^o
25^o	23.0^o	20.0^o	21.50^o	3.0^o
30^o	28.2^o	23.7^o	25.95^o	4.5^o
33^o	30.0^o	24.7^o	27.35^o	5.3^o

5.5 Design Analysis

When all the steering system's parts are completely fabricated, it has been assembled to complete vehicle. The actual result as shown in Table 5.12 and Figure 5.9 below

Table 5.12: The actual result

Steering angle, δ_H	Outer angle, δ_o	Inner angle, δ_i	Mean wheel angle. δ_{mean}	Difference between angle for inner wheel and outer wheel, $\lvert\delta_i - \delta_o\rvert$
0^o	0^o	0^o	0^o	0^o
5^o	3^o	3^o	3.00^o	0^o
10^o	7^o	6^o	6.50^o	1.0^o
15^o	11^o	9^o	10.00^o	2.0^o
20^o	16^o	12^o	14.00^o	4.0^o
25^o	20^o	15^o	17.50^o	5.0^o
30^o	25^o	19^o	22.00^o	6.0^o
35^o	30^o	23^o	26.50^o	7.0^o

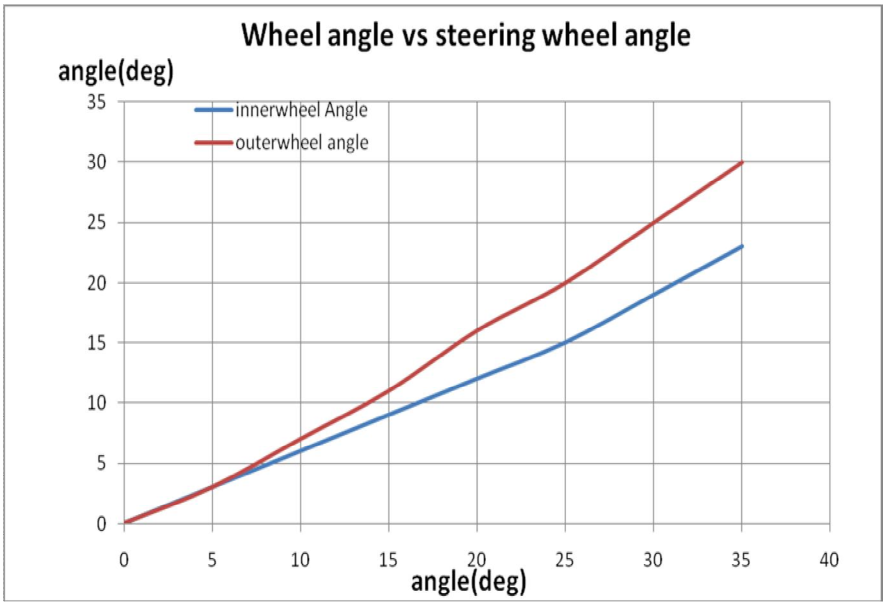


Figure 5.9: Graph for the actual result of the front steer wheel angle versus steering wheel angle

Steering ratio=1.32 (see appendix for calculation)

Turning radius=4.0m

Table 5.13: Comparison between the simulation result with the actual result

	Simulation	**Actual**
Inner wheel angle, δ_i	24.7^o	23.0^o
Outer wheel angle, δ_o	30.0^o	30.0^o
Steering wheel angle, δ_H	31.8^o	35.0^o
Steering ratio	1.16	1.32
Turning radius, m	4	4

There are differences between the actual and simulation result as shown in Table 5.13 above due to several factors such as misalignment of the matching part, tolerances of each parts and limitation of manufacturing process.

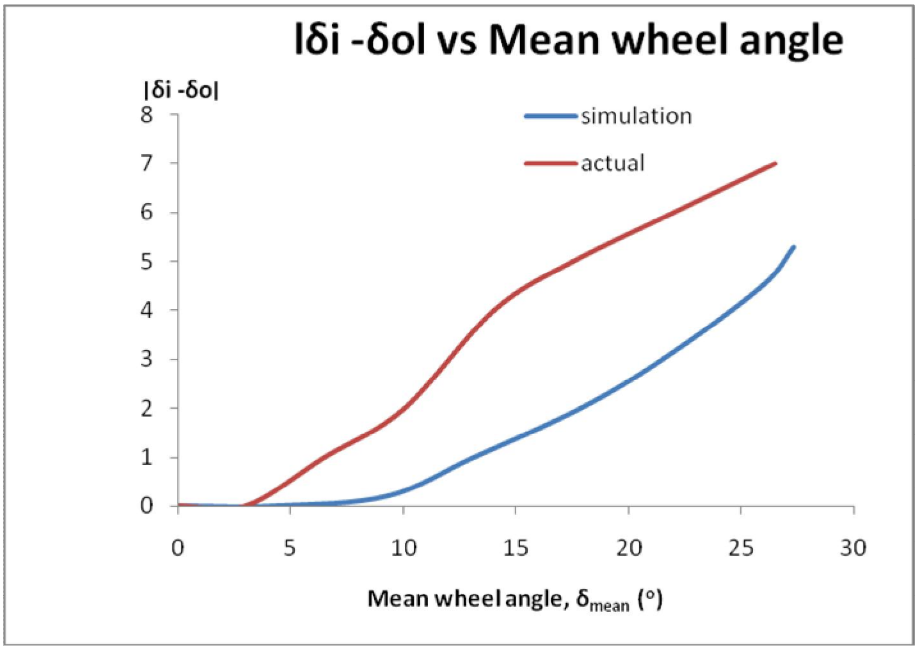


Figure 5.10: Toe-out angle as a function of the mean steering angle based on simulation result and actual result

By comparing Figure 5.10 with Figure 3.5 in theory, the pattern of graph is similar. It can be concluded that the final design of the steering system is validate.

Base on the Table 5.11 and 5.12, the outer wheel angle is greater than the inner wheel angle. The result is totally different with the true Ackermann steering geometry.

At low speeds when the tires have minimal tire shear losses on dry, clean pavement, the design causes the car experience understeer. This scenario would push the front of the car away from the desired path. In this situation, the inner tire contributes to this push similarly to a centrifugal force.

At high lateral accelerations, the design is beneficial since the inner and outer wheel still have lateral grip. The inner wheel has also surpassed the maximum slip angle of grip assuming the outer wheel is already at the optimum slip angle. The outer wheel (which currently has more loads due to weight transfer) is at the optimum slip angle and the inner wheel is at a lower slip angle with fewer grips. As the result, it allows the inner wheel to have grip but less than the outer wheel and also decreasing the effects of understeer during taking the turn.

CHAPTER 6
CONCLUSION AND RECOMMENDATION

In order to design and select steering system as light and simple as possible and meet the sufficient turning radius to be participated in Shell Eco-marathon 2010, the comparison and information about the teams that already participated in Shell Eco-marathon is needed. The experience from team such as Pac-Car ETH Zurich from Switzerland and Cal Poly Supermileage from California Polytechnic State University, USA become guidance in designing the steering system. Their experience in the selection of steering system is very useful.

Generally, the specifications of steering system are light, the steering geometry should not get affect by bad road conditions, equipped with self-centering, easily operateable and front wheel steer. For the purpose of this project, the Product Design Specification(PDS) was determined in order to achieve the objective of this project. The PDS of the steering system are the maximum angle for inner wheel and outer wheel is high as possible, small turning radius, small steering ratio, follows true Ackermann steering geometry, the difference between angle for inner wheel and outer wheel small as possible and lastly, the steering system should be light to turn the wheel.

The material selection for steering system components has been done by using weight property index method. From this method, the aluminium alloy is most reliable to use in fabricating the tie rod and the steering column and stainless steel is most reliable to use in fabricating the spindle. Four design of steering system were proposed in the beginning, after do some analysis, simulation in ADAMS and weight property index, design 1 was selected. This steering system has used simple link mechanism and the mechanism almost similar with go-kart steering system. The design does not consist of the tie rod arm and the tie rod is connected directly to spindle by using ball joint.

In the optimization step, some of the parameters have been changed when designing the steering column in order to ensure that the steering system meet the product design specification (PDS). The optimize design has been simulated in ADAMS again. The result of simulation indicates the inner wheel angle and outer wheel angle of final design of steering system are 24.7^{o} and 30.0^{o} at steering wheel angle of 31.8^{o}. The turning radius is 4m and the steering ratio is 1.16.

Then, the steering system has been fabricated based on the modeling in CATIA. The actual result indicates the inner wheel angle and outer wheel angle of final design of steering system are 23.0^{o} and 30.0^{o} at steering wheel angle of 35.0^{o}. The turning radius is 4m and the steering ratio is 1.32.

Since the final design does not followed true Ackermann steering geometry, the car has experienced understeer that this scenario would push the front of the car away from the desired path. For the improvement, the steering system should be designed properly that will follow true Ackermann steering geometry. The true Ackermann steering geometry is beneficial as the tires are in almost a perfect situation of minute slip angle especially at low speeds when the tires have minimal tire shear losses on dry clean and pavement.

Although the design does not follow Ackermann steering geometry, the steering system can be used safely in order to be participated in Shell Eco-Marathon 2010.

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APPENDIX I

FYP Gantt Chart

FYP I Gantt Chart

Week Number / Activities/Milestone	1	2	3	4	5	6	7	8	9		10	11	12	13	14	15
Selection of topic Brain Storming of ideas																
Develop scope of project																
Develop PDS																
Submission preliminary report				21/8												
Produce conceptual decomposition,																
Develop and validate concept design																
Develop product architecture			1 week break							Mid-sem break						
Submission of progress report								10/9								
Seminar								15/9								
Develop CAD design																
Validate design by using ADAMS																
Selection of design and materials																
Submission of interim report final draft																5/11
Oral presentation																

FYP II Gantt Chart

Week Number / Activities/Milestone	1	2	3	4	5	6	7		8	9	10	11	12	13	14	15
Produce detail drawing																
Validate final design: -Simulation -PDS																
Submission of progress report 1																
Clarification of machining process, tolerance																
Preparation of material, machine, tool, equipment																
Manufacture																
Installation the steering system to complete car								Mid-sem break								
Submission of progress report 2																
Seminar																
Validate/comparison with PDS and simulation result																
Poster exhibition																
Submission of dissertation final draft																
Oral presentation																
Submission of dissertation(Hard bound)																

APPENDIX II

Sample of Calculation

R_s=4m, l=2m

$$R_s = \frac{l}{\sin\delta_o}$$

$$\delta_o = \sin^{-1}\left(\frac{l}{R_s}\right)$$

$$= \sin^{-1}\left(\frac{2}{4.0}\right)$$

$$= 30^o$$

Design 1

Wheelbase, l=2m

Outer wheel angle, $\delta_o = 30^o$ inner wheel angle, $\delta_i = 18.8^o$

Steering wheel angle, $\delta_H = 48^o$

$$\delta_m = \frac{\delta_o + \delta_i}{2}$$

$$= \frac{30^o + 18.8^o}{2}$$

$$= 24.4^o$$

$$i_s = \frac{\delta_H}{\delta_m}$$

$$= \frac{48^o}{24.4^o}$$

$$= 1.97$$

$$|d_i - d_o| = |18.8^o - 30^o|$$

$$= 11.2^o$$

Final design

Outer wheel angle, $\delta_o = 30^o$ inner wheel angle, $\delta_i = 23^o$

Steering wheel angle, $\delta_H = 35^o$

$$\delta_m = \frac{\delta_o + \delta_i}{2}$$

$$= \frac{30^o + 23^o}{2}$$

$$= 26.5^o$$

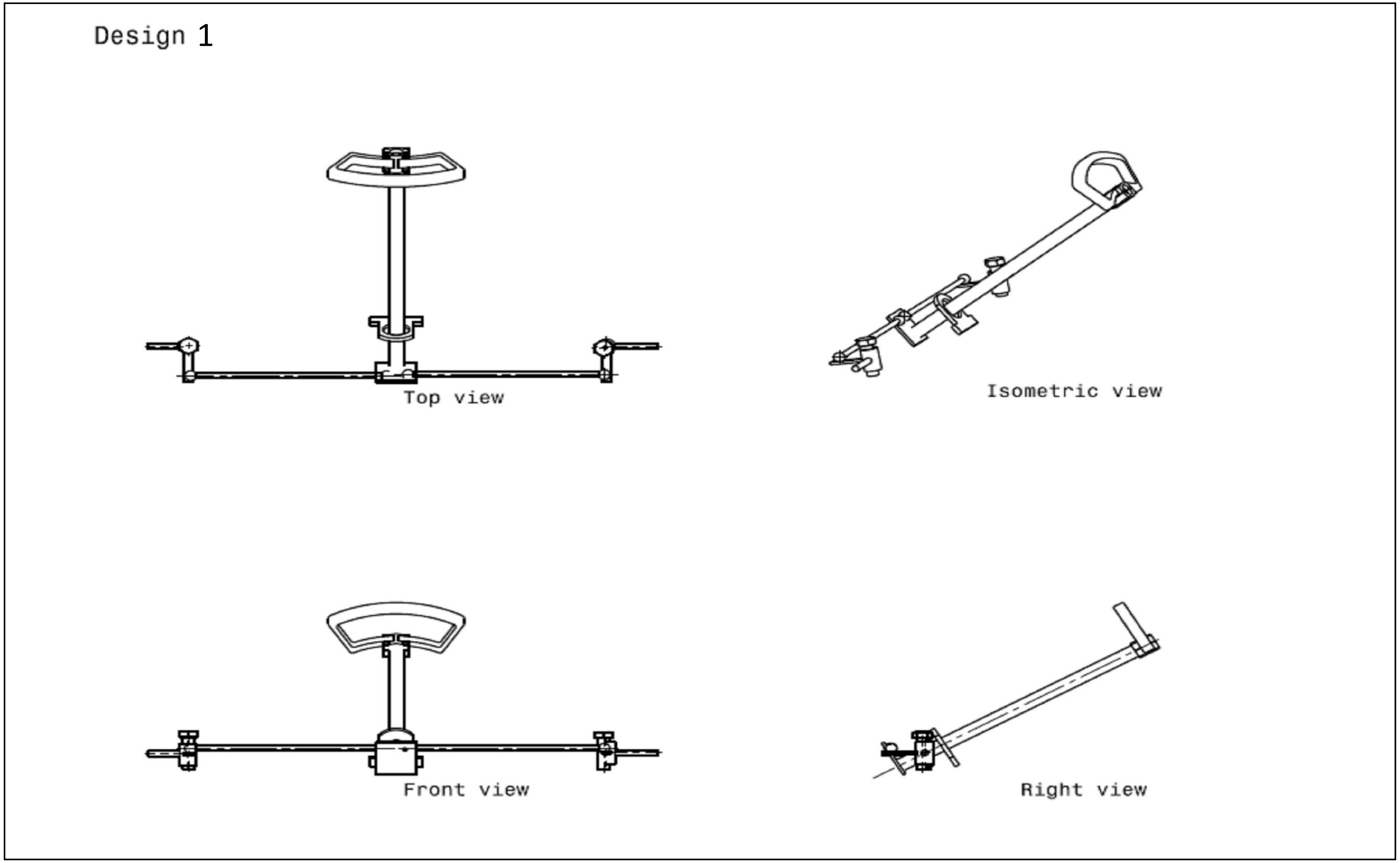
$$i_s = \frac{\delta_H}{\delta_m}$$

$$= \frac{35.0^o}{26.5^o}$$

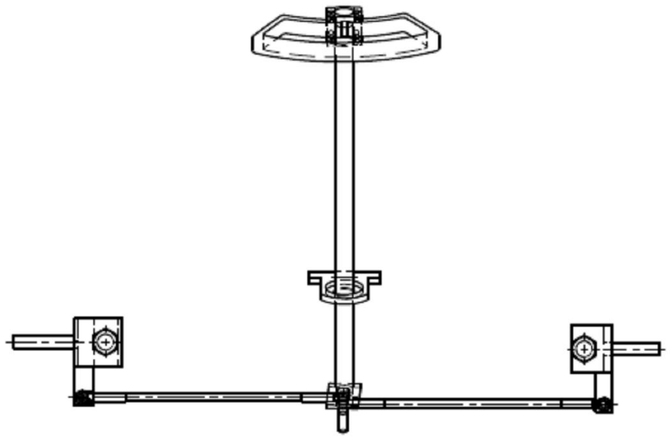
$$= 1.32$$

APPENDIX III

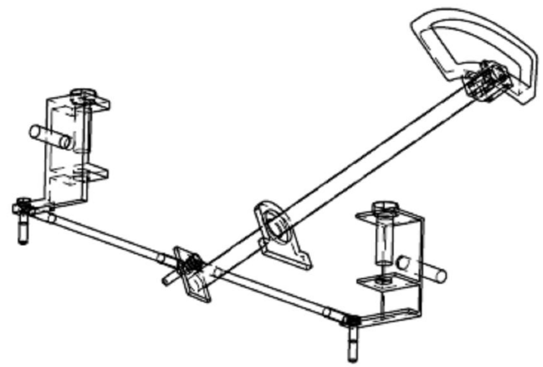
Alternative Designs



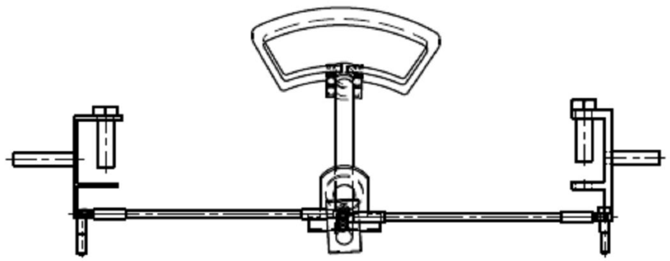
Design 2



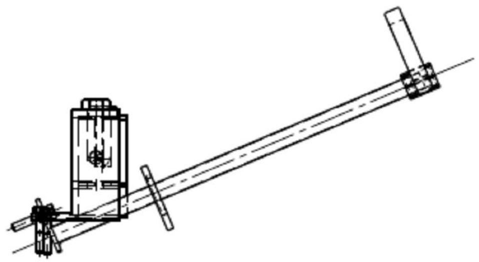
Top view



Isometric view

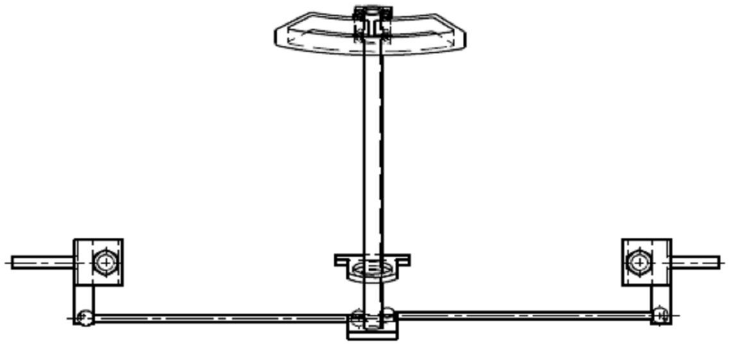


Front view

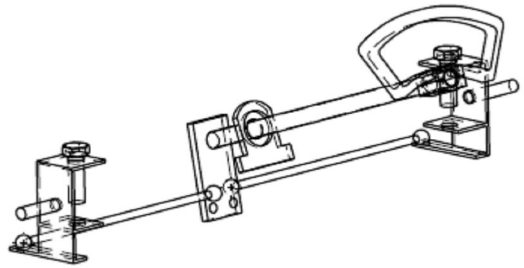


Right view

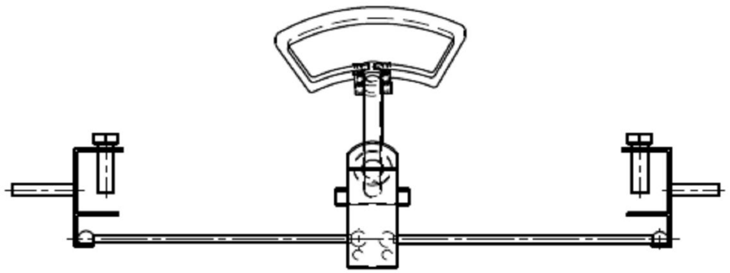
Design 3



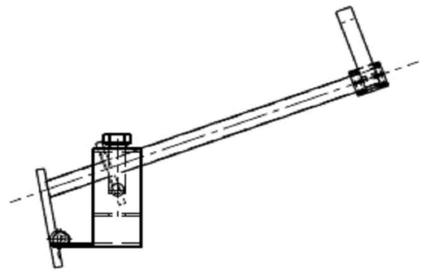
Top view



Isometric view

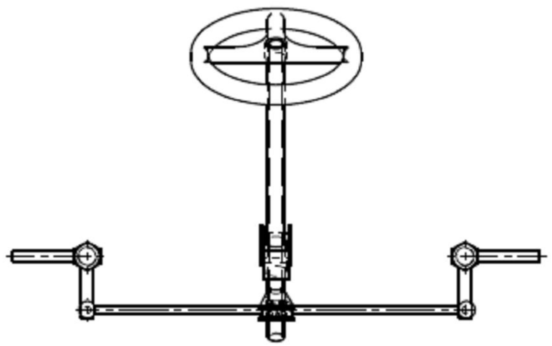


Front view

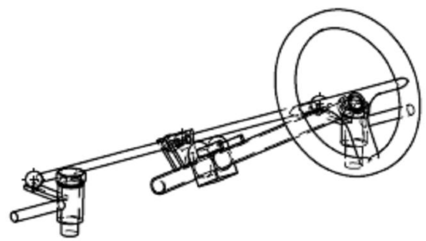


Right view

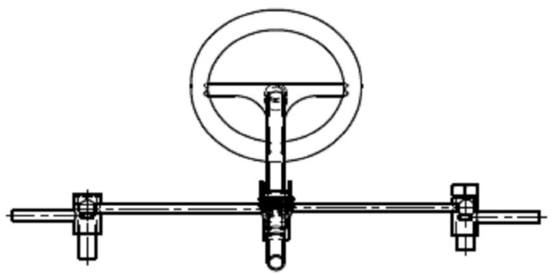
Design 4



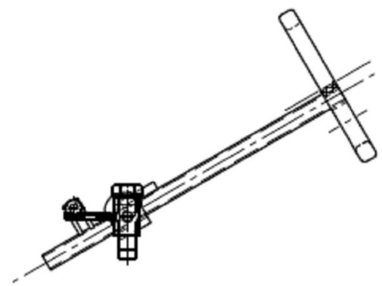
Top view



Isometric view



Front view

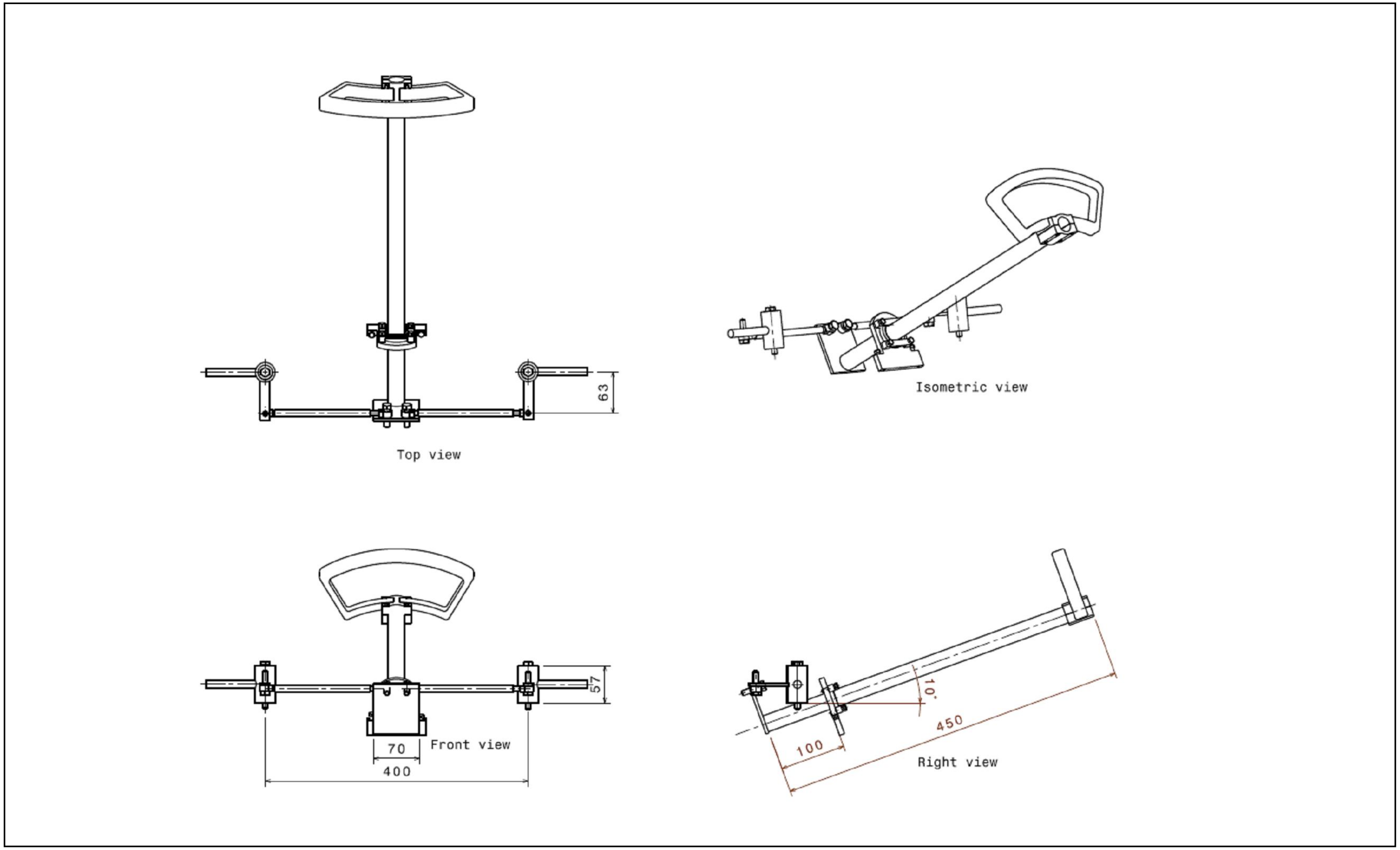


Right view

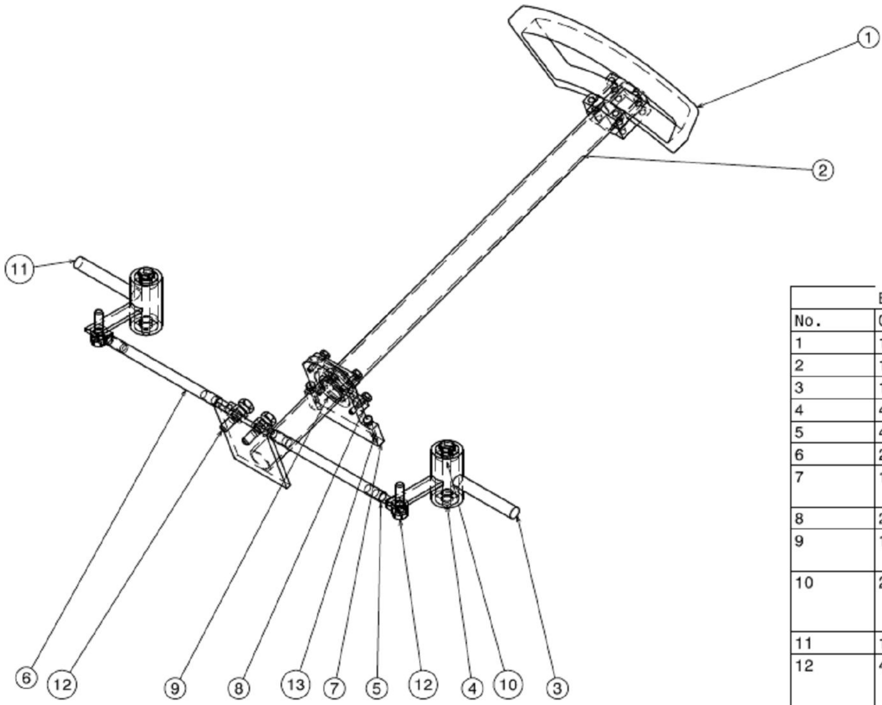
APPENDIX IV

Final Design

Projection View of Final Design



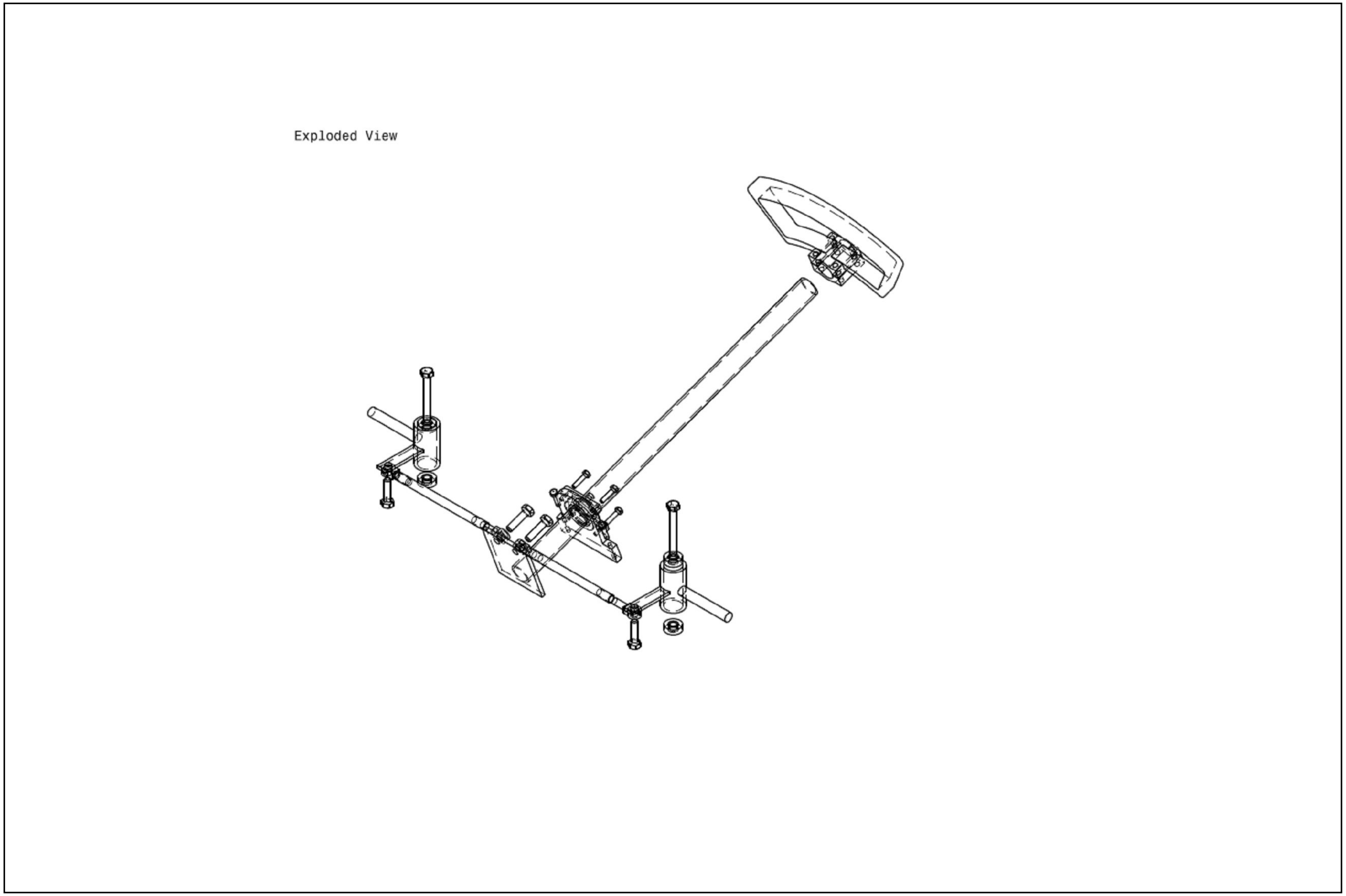
Final Design with Label



Bill of Material: Product2

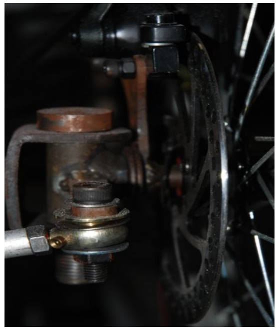
No.	Quantity	Part Name	Material
1	1	Steering Wheel	Plastic
2	1	Steering Column	Aluminium
3	1	Left Spindle	Steel
4	4	Bearing 6001 RS	Steel
5	4	Ball Joint	Steel
6	2	Tie Rod	Aluminium
7	1	Steering Column Holder	Aluminium
8	2	Bearing Holder	Aluminium
9	1	Bearing Steering Column 6005 C3	Steel
10	2	ISO 4018 SCREW M8x65 STEEL GRADE C HEXAGON HEAD	Steel
11	1	Right Spindle	Steel
12	4	ISO 4018 SCREW M8x30 STEEL GRADE C HEXAGON HEAD	Steel
13	6	ISO 4018 SCREW M5x20 STEEL GRADE C HEXAGON HEAD	Steel

Exploded View of Final Design



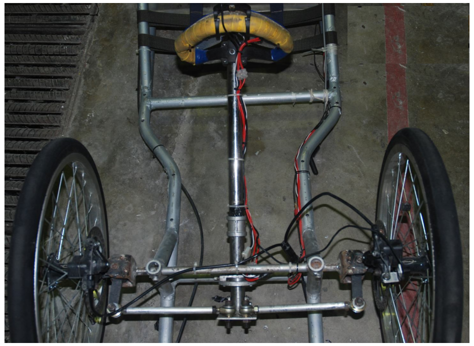
APPENDIX V

Pictures of Real Design of Steering System



Front view of spindle

Bearing holder



Steering system

Top view of spindle

Tie rod